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# A WEIGHTING METHOD TO DETERMINE THE IMPACT OF VOLUMETRIC EFFICIENCY ON THE THERMODYNAMIC EFFICIENCY OF A COMPRESSOR

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## ABSTRACT

It is a generally accepted fact that the volumetric efficiency of any given compressor plays a very important role in the overall thermodynamic efficiency of the compressor. In this paper a novel performance analysis technique has been developed that allows volumetric efficiency loss mechanisms to be considered as significant contributors to the thermodynamic losses occurring within a compressor. While the 'isentropic efficiency' of the compressor remains unchanged, the effect of the procedure is to negatively weight the processes that cause both volumetric and thermodynamic losses (i.e. processes occurring on the suction side). The new methodology is compared to an exergy analysis.

## NOMENCLATURE

b	Specific flow exergy ( $b=h-T_0.s$ ) (J/kg)	ch	working volume (chamber) of the compressor
$C_p$	Specific heat capacity at constant pressure (J/kgK)	cond	condenser
h	Specific enthalpy (J/kg)	dsch	conditions at compressor outlet (discharge)
I	Exergy destruction (J)	dthr	discharge throttling
Loss	Instantaneous loss in mass flow rate (kg/s)	fr	mechanical friction
$\dot{m}$	Mass flow rate (kg/s)	HT	heat transfer
n	Revolutions per sec (Hz)	ideal	ideal efficiency (mass induced/expansion)
P	Total Pressure (N/m <sup>2</sup> )	loss	total losses (mass flow rate)
Q	Heat transfer (J)	leak	leakage
R	Specific gas constant (J/kgK)	max	maximum
s	Specific entropy (J/kgK)	min	minimum
T	Temperature (K)	mix	due to mixing of gas or oil masses
t	Time (sec)	0	Ambient conditions
V	Volume (m <sup>3</sup> )	s	isentropic
W	Work (J)	sthr	suction throttling
$\gamma$	Ratio of specific heat capacities (dimensionless)	suc	conditions at compressor inlet (suction)
$\Delta$	Finite difference, Change in..	thr	throttling
$\delta$	Differential operator	time	associated with timing defect
$\eta$	Isentropic efficiency (dimensionless)	tot	total
$\eta_v$	Volumetric efficiency (dimensionless)		
$\theta$	Crank angle		
$\rho$	Gas density (kg/m <sup>3</sup> )		

## 1.0 INTRODUCTION

A refrigeration/air conditioning system has two primary roles as a heat pump. These are..

Subscripts/Superscripts:

actual measured (mass flow rate)  
cl clearance volume

a) to create heat flow ( $\dot{Q}$ )  
b) to reverse the natural

direction of the heat flow, (i.e. from low to high temperature), by the application of mechanical work.

Therefore the two primary measurements that are of interest are the **heat flow rate** and the **work input**. In vapour compression refrigeration cycles all the components contribute to the system performance, however narrowing down the analysis to the compressor only, it will be shown how a) the volumetric efficiency and b) the isentropic efficiency of the compressor can approximate these primary measurements.

Individually the efficiencies for flow and for work input have been studied in the literature, references to these researches are given in [1], [3], [4]. However few authors have investigated the interaction that these two effects have on each other [6]. It is the purpose of this paper to study this relationship.

Why is there an interest in the inter-relationships between the volumetric efficiency and thermodynamic efficiency of a compressor? In the course of compressor design and manufacture there have been several instances where the inter-relationships have proven to be of design interest.

- In fixed displacement compressors, there is generally a strong correlation between those with poor volumetric efficiency and those that have low thermodynamic efficiencies, however this is not always true.

- Also, variable capacity/displacement compressors are becoming more common in automotive applications. As the volumetric efficiencies decrease to match capacity, the discharge temperature can become a significant design issue (for reference see figure 3).

## 2.0 MODEL DESCRIPTION

In the following section a model is proposed that creates an inter-relationship between the two efficiencies.

### 2.1) Efficiency Measures

A measure is required that will correlate to the heat flow rate. In the vapour-compression refrigeration

cycle, the heat flow rate can be approximated by the mass flow rate of the refrigerant. The volumetric efficiency of the compressor is a good choice, as it relates the mass flow rate to the design parameters of the compressor (i.e. swept volume).

$$\eta_v = \dot{m}_{\text{actual}} / (\rho_{\text{suc}} \cdot (V_{\text{max}} - V_{\text{min}}) \cdot n) \quad \text{Eqn. (1)}$$

A measure is required that will correlate to the work input and also to the discharge temperature. The isentropic efficiency is the best gauge for this. The work input can be referenced against a thermodynamic ideal, and the discharge temperature can be related directly to it, (if oil flow is low and there is not considerable heat transfer between the compressor and the ambient air). Isentropic efficiency is defined as follows,

$$\eta = \dot{m} \cdot \Delta h_s / W \quad \text{Eqn. (2)}$$

The individual losses that contribute to the decrement in these two efficiencies, can now be considered.

#### 2.1.1) Shaft Power Losses

The total power input is equal to the total exergy transfer to the working fluid as it passes through the compressor plus the summation of all the exergy destruction (loss) mechanisms occurring within the compressor [1], [3].

$$W = \dot{m} \cdot (b_{\text{dsch}} - b_{\text{suc}}) + \dot{I}_{\text{tot}} \quad \text{Eqn. (3)}$$

To insert this into equation (2) it is preferable to use the isentropic enthalpy change occurring within the compressor,

$$W = \dot{m} \cdot \Delta h_s + \dot{I}_{\text{cond}} + \dot{I}_{\text{tot}} \quad \text{Eqn. (4)}$$

This was achieved by adding another loss term to the equation. This extra term ( $\dot{I}_{\text{cond}}$ ) represents the loss due to heat transfer, that occurs in the condenser, bringing the discharge gas to isentropic discharge gas temperature. As generally the exergy of the discharge gas is never normally recovered to generate useful work, there is no foreseeable problem

associating this exergy destruction as a compressor loss and not as a condenser loss [2], [5].

$$\dot{I}_{\text{tot}} = \sum \dot{I}_{\text{thr}} + \sum \dot{I}_{\text{fr}} + \sum \dot{I}_{\text{mix}} + \sum \dot{I}_{\text{HT}} \quad \text{Eqn. (5)}$$

$$\dot{I}_{\text{cond}} = \dot{m} \cdot (b_{\text{dsch}} - b_{\text{suct}}) - \dot{m} \cdot \Delta h_s \quad \text{Eqn. (6)}$$

### 2.1.2) Displacement Utilization Losses

The losses that reduce the displacement utilization efficiency (volumetric efficiency) of a compressor can be accounted for by subtracting the loss generating mechanisms [4], as follows:

$$\dot{m} = \dot{m}_{\text{ideal}} - \dot{m}_{\text{loss}} \quad \text{Eqn. (7)}$$

where,

$$\dot{m}_{\text{loss/cyc}} = \oint \text{Loss}_{\text{HT}} \cdot \delta t + \oint \text{Loss}_{\text{thr}} \cdot \delta t + \oint \text{Loss}_{\text{ci}} \cdot \delta t + \oint \text{Loss}_{\text{leak}} \cdot \delta t + \oint \text{Loss}_{\text{time}} \cdot \delta t \quad \text{Eqn. (8)}$$

### 2.2) Weighting Factors

Combining the isentropic efficiency (eqn.2), the loss generating mechanisms that reduce the isentropic efficiency (eqn.4) and the definition of volumetric efficiency (eqn.1), the isentropic efficiency is redefined as follows.

$$\eta = \eta_V \cdot \dot{m}_{\text{ideal}} \cdot \Delta h_s / (\eta_V \cdot \dot{m}_{\text{ideal}} \cdot \Delta h_s + \dot{I}_{\text{tot}} + \dot{I}_{\text{cond}}) \quad \text{Eqn. (9)}$$

Equation (9) states the isentropic efficiency is a function of both the volumetric efficiency and the exergy destruction rates. Fig. 1 shows a surface contour plot for how these two components combine to reduce the isentropic efficiency of a compressor.

Differentiating equation (9) yields the following relation for the incremental effect that these effects have on the overall thermodynamic performance. (For this derivation, it is assumed that  $\dot{I}_{\text{cond}}$  is a constant and not a function of  $\eta_V$  or  $\dot{I}_{\text{tot}}$ , for a large portion of the map this is approximately true, refer to Fig. 2).

$$\begin{aligned} d\eta &= (\delta\eta/\delta\eta_V) \cdot d\eta_V + (\delta\eta/\delta\dot{I}_{\text{tot}}) \cdot d\dot{I}_{\text{tot}} \\ &= [((\dot{I}_{\text{tot}} + \dot{I}_{\text{cond}}) \cdot d\eta_V - \eta_V \cdot d\dot{I}_{\text{tot}}) \cdot \dot{m}_{\text{ideal}} \cdot \Delta h_s] / (\eta_V \cdot \dot{m}_{\text{ideal}} \cdot \Delta h_s + \dot{I}_{\text{tot}} + \dot{I}_{\text{cond}})^2 \end{aligned} \quad \text{Eqn. (10)}$$

It can be readily seen from this equation that if the volumetric efficiency is improved or if the exergy destruction rate is reduced, the overall isentropic efficiency will be improved, (as expected). Graphically, the derivative above is the slope on Fig.1, for a given change in volumetric efficiency and/or the exergy destruction rate. This equation forms the basis of the analysis following.

## 3.0 MODEL APPLICATION

In this section, the existing and the newly proposed analysis techniques are applied to a compressor simulation program. The predictions of the performance changes are then compared with actual changes in the simulation program.

### 3.1) Reciprocating Compressor Model

To illustrate the technique a reciprocating compressor simulation is used. The model has been described in previous work [3]. The results are shown in Table 1a. As has been mentioned above, techniques exist to analyse the efficiencies independently. Included in tables 1b and 1c therefore are the results of an exergy destruction analysis and a volumetric efficiency analysis.

Three special cases shall be considered to find their relative effects on the isentropic efficiency. An effect that reduces the volumetric efficiency and which also destroys exergy is examined, such as suction throttling, piston seal leakage, heat transfer to the suction gas as in table 1b. An effect that only reduces the volumetric efficiency but does not destroy exergy is considered, an example is re-expansion of the trapped volume in a reciprocating compressor, as shown in table 1b. Also an effect that does not reduce the volumetric efficiency of a compressor, but which does destroy exergy is examined, such as heat transfer in the metal housing, discharge valve throttling, heat transfer to ambient, heat transfer on

the discharge side, friction and external heat transfer occurring in the condenser, table 1b.

### 3.1.1) Case 1: Suction throttling

Suction throttling affects the isentropic efficiency by two mechanisms. Primarily it affects the efficiency due to the thermodynamic irreversibility of the process itself, the flowing of refrigerant from a higher pressure region to a lower pressure region. Secondly, it affects the isentropic efficiency due to the fact that it reduces the mass flow through the compressor (volumetric efficiency).

It can be shown that the exergy destruction rate associated with suction throttling is as follows [3], (table 1b shows this loss to be 76W),

$$\dot{I}_{thr-suc} = \dot{m} \cdot T_0 \cdot ((s_{ch} - s_{suc}) - (h_{ch} - h_{suc}) / T_{ch}) \cdot d\theta / 2\pi \quad \text{Eqn. (11)}$$

It can also be shown that the loss in the mass flow rate due to suction throttling is as follows [4], this loss was calculated to be 1.429 g/s (note: this also includes 'timing' losses),

$$\text{Loss}_{thr-suc} = \int ((P_{suc} - P_{ch}) \cdot \dot{V} / C_p \cdot T_{suc}) \cdot d\theta / 2\pi \quad \text{Eqn. (12)}$$

and therefore..

$$\Delta\eta_{Vsthr} = \text{Loss}_{thr-suc} / \dot{m}_{ideal} \quad \text{Eqn. (13)}$$

This information is employed to calculate the **real** effect of the suction throttling process on the isentropic efficiency. The values are inserted into their respective functions as proposed in equation (10). (Note: by **real** it is meant that we are including the effect on volumetric efficiency as being a real component of the isentropic loss generation process).

$$\Delta\eta_{tot-sthr} = ((\dot{I}_{tot} + \dot{I}_{cond}) \cdot \Delta\eta_{Vsthr} - \eta_V \cdot \dot{I}_{thr-suc}) \cdot \dot{m}_{ideal} \cdot \Delta h_s / (\eta_V \cdot \dot{m}_{ideal} \cdot \Delta h_s + \dot{I}_{cond} + \dot{I}_{tot})^2 \quad \text{Eqn. (14)}$$

When calculated, this will yield an efficiency drop of 8.57%. However this is before any weighting has been applied, which shall be explained later.

### 3.1.2) Case 2: Re-expansion volume

The re-expansion volume in a reciprocating compressor is the volume that is trapped when the compressor is at the top dead centre. After top dead centre, the gas at discharge pressure expands to suction pressure. Therefore it takes a portion of the expansion stroke before the pressure drops below suction pressure, allowing suction to occur. During this period the compressor has lost it's potential to effectively use this displaced volume to induce flow into the compressor. While the process itself does not require exergy destruction to occur, (an isentropic expansion is feasible), the effect on the volumetric efficiency ultimately will contribute to the loss in isentropic efficiency. This case is chosen to demonstrate a process that does not destroy exergy but which ultimately reduces the isentropic efficiency of a compressor.

The mass lost due to the re-expansion process is as follows, (in table 1c this loss was 1.485 g/sec),

$$\text{Loss}_{cl} = ((P_{dsch} \cdot V_{cl}^\gamma / P_{suc})^{1/\gamma} - V_{cl}) \cdot P_{suc} / (R \cdot T_{suc}) \cdot n \quad \text{Eqn. (15)}$$

Therefore the isentropic efficiency loss due to the process follows from this.

$$\Delta\eta_{tot-cl} = (\dot{I}_{tot} + \dot{I}_{cond}) \cdot \Delta\eta_{Vcl} \cdot \dot{m}_{ideal} \cdot \Delta h_s / (\eta_V \cdot \dot{m}_{ideal} \cdot \Delta h_s + \dot{I}_{cond} + \dot{I}_{tot})^2 \quad \text{Eqn. (16)}$$

When calculated, this mechanism now accounts for 2.34% of the loss in the isentropic efficiency. Again this is before any weighting has been applied.

### 3.1.3) Case 3: Discharge Throttling

Discharge throttling is the third example of what can occur in a compressor that will affect the isentropic efficiency. In this case, the throttling destroys exergy in an identical manner to the suction

throttling process. However, unlike in the suction throttling case there is no associated reduction in the volumetric efficiency of the compressor. The exergy destruction associated with discharge throttling is as follows (95 W, table 1b),

$$\dot{I}_{thr-dsch} = \dot{m} \cdot T_0 \cdot ((s_{dsch} - s_{sch}) - (h_{dsch} - h_{sch}) / T_{dsch}) \cdot d\theta / 2\pi$$

Eqn. (17)

However, according to the new scheme developed, the actual contribution of this to the overall isentropic efficiency is actually less than would be assigned by an exergy analysis,

$$\Delta\eta_{tot-dthr} = - \eta_v \cdot \dot{I}_{thr-dsch} \cdot \dot{m}_{ideal} \cdot \Delta h_s / (\eta_v \cdot \dot{m}_{ideal} \cdot \Delta h_s + \dot{I}_{cond} + \dot{I}_{tot})^2$$

Eqn. (18)

The contribution to the efficiency loss is therefore 7.86%, before weighting.

### 3.1.4) Weighting Procedure

The use of equation (10) is dependant on there being only small deviations around the existing performance of the compressor. Therefore the loss contributions to the isentropic efficiency cannot be arithmetically summed to give the overall percentage loss in the isentropic efficiency of the compressor. However, by using the existing performance as the baseline condition, equation (10) can be used to **weight** the relevant factors that contribute to the overall efficiency.

In this example, summing all the loss percentages yields an overall efficiency loss of 29.83%. However the isentropic efficiency is actually 61.69%. Consequently, scaling up the loss percentages by 38.31/29.83, the losses will sum correctly. Therefore, for example, the suction throttling loss according to the proposed method would now be 11.0% or 82 W compared to an exergy analysis calculation of 76 W. The discharge throttling losses on the other hand, went from an exergy destruction loss of 95 W, to only being accountable for 75 W. It can be seen how this new technique strongly emphasizes processes occurring on the suction side.

### 3.2) Model Verification (Loss prediction)

It was the purpose of the technique to predict how changes, when implemented, will affect the performance. Design changes to improve the performance of the compressor will be based on the detailed component analyses in tables 1b and 1c.

In table 1b, the columns represent..

- the isentropic efficiency losses by exergy techniques.
- the isentropic efficiency losses by applying the new weighted analysis technique.
- the volumetric efficiency losses.

Using this table, it can be seen that (according to an exergy analysis) there is no shaft power lost directly due to re-expansion of the gases in the clearance volume. There is however a 7.8% loss in the volumetric efficiency. The new method does predict a 3% loss in the isentropic efficiency due to the clearance volume, contrary to the exergy analysis.

Table 2 represents the case where the clearance volume was reduced by half in the simulation. As was expected the volumetric efficiency improved from 78.9% to 83%, a 4.1% improvement (or half the percentage loss of 7.8% predicted for the full clearance volume). The improvement in the isentropic efficiency however was only 0.3% (from 61.7% to 62%), not the 1.5% predicted. Part of the failure of the method to predict the improved performance can be sourced back to the use of equation 10. In using this equation, the fact that other exergy destruction mechanisms would change as a result of the intended design change was ignored. This is most graphically seen in the fact that the discharge valve losses tended to compensate almost 100% for improvement due to increased flow.

Table 3 shows what would occur if the base design were to have half the friction component that it presently has. As can be seen in the base design level, friction accounted for 6.2% of the overall losses within the compressor, therefore if the friction were to be reduced by

approximately half, the expected improvement in isentropic efficiency should be of the order of 3.1%. In fact the improvement was only 2.43%. Compare this with the new method which predicted that the performance improvement would be 2.45% ( $4.9\%/2$ ) and we can see that the agreement is much closer.

#### 4.0 DISCUSSION

The previous two examples helped show the strengths and weaknesses of the new method. In the first case the efficiency improvement predicted was not accurate. The primary reason given for the lack of agreement was the fact that secondary effects were found to play an important role in the outcome. This is always a problem in system analysis and suggests that the base equation 10 should have modifiers to account for important secondary effects. This is beyond the context of this paper however. In the second example the method was good at predicting the efficiency improvement.

Analysing these tables again it can be noticed how the proposed new method weights the losses occurring on the suction side at the expense of those on the discharge side. This is an important outcome of the analysis and helps give a sound fundamental basis to concentrating more effort on suction side processes, as these processes are fundamental to everything else that occurs to the gas within the compressor.

Another positive outcome is in the design of variable capacity compressors. In these compressors the principal concern is to decrease the volumetric efficiency while maintaining the discharge temperatures at reasonable values. Different design concepts can therefore be evaluated more effectively by the use of equation 10.

#### 5.0 CONCLUSIONS

In this paper a new way of looking at the isentropic efficiency of a compressor has been proposed. The technique is useful in that it combines the two primary functions of the compressor, the mass flow and compression processes, in a logical manner. It has been long known that mass flow does affect the isentropic

efficiency of a compressor, however a relation between the two has not been proposed that can be applied to the detail that this new technique offers.

#### ACKNOWLEDGEMENTS

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#### REFERENCES

- 1] McGovern, J.A., Harte, S.A., and Strikis, G.V., 'Real Gas Performance Analysis of a Scroll or Rolling Piston Compressor Using Exergy Techniques', Proc. of the 1994 International Compressor Engineering Conference at Purdue, Ed. Soedel, 1994, Vol.1, pp.193-198.
- 2] Xu, X. 'Exergy Analysis on a Vapour Compression System using R12, R134a and R290 as Refrigerants', Proc. of the 1992 International Refrigeration Conference at Purdue, Ed. Tree and Braun, 1992, Vol.1, pp.231-240.
- 3] McGovern, J.A. and Harte, S.A., 'An Exergy Method for Compressor Performance Analysis', International Journal of Refrigeration, Vol. 18, No. 6, pp421-433, 1995.
- 4] Harte, S.A., Strikis, G.V., and McGovern J.A., 'Instantaneous Displacement Utilization Analysis of Positive Displacement Compressors', ASME International Mechanical Engineering Congress, 1995, AES-Vol. 34, pp. 165-172.
- 5] Szargut, J., Morris, D.R., and Steward, F.R., 'Exergy Analysis of Thermal, Chemical, and Metallurgical Processes', Hemisphere Publishing Corp., 1988.
- 6] Pandeya, P.N. and Soedel, W., 'A Generalised Approach Towards Compressor Performance Analysis', Proc. of the 1978 Purdue Compressor Technology Conference, Ed. Hamilton, 1978, pp. 135-143.

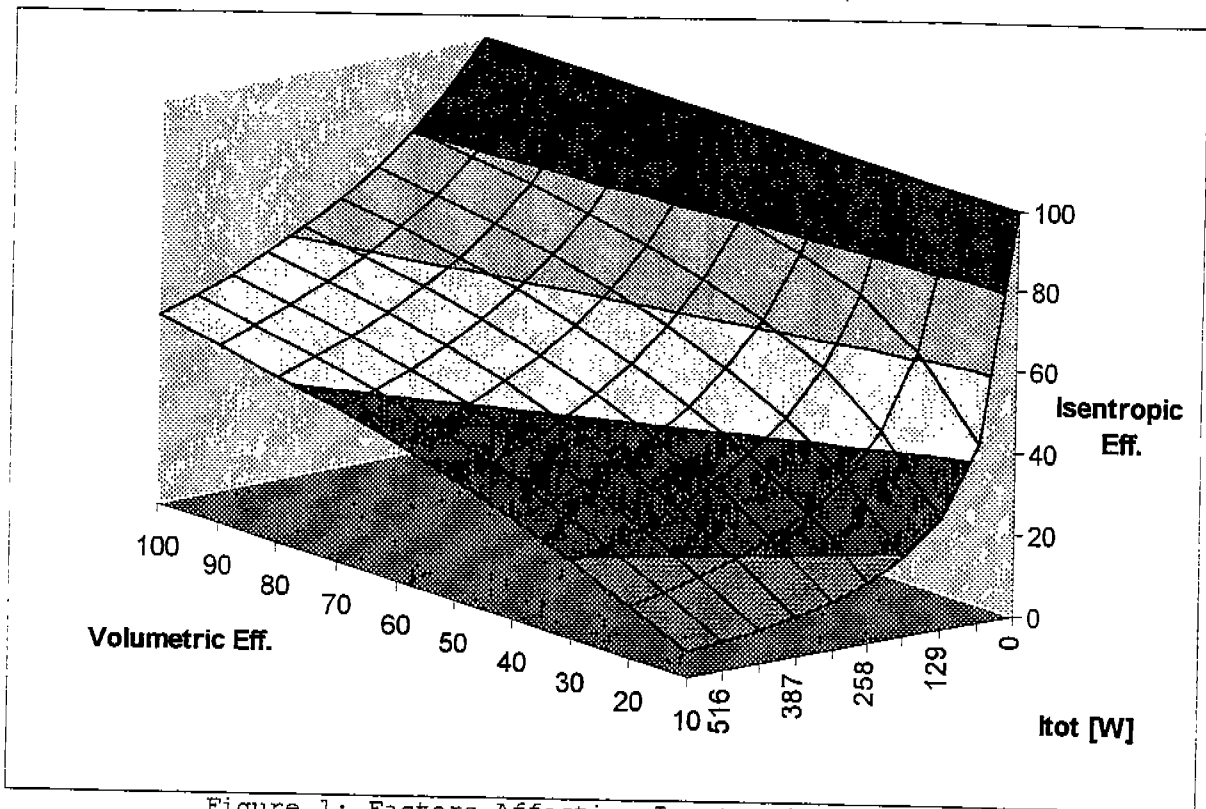


Figure 1: Factors Affecting Isentropic Efficiency

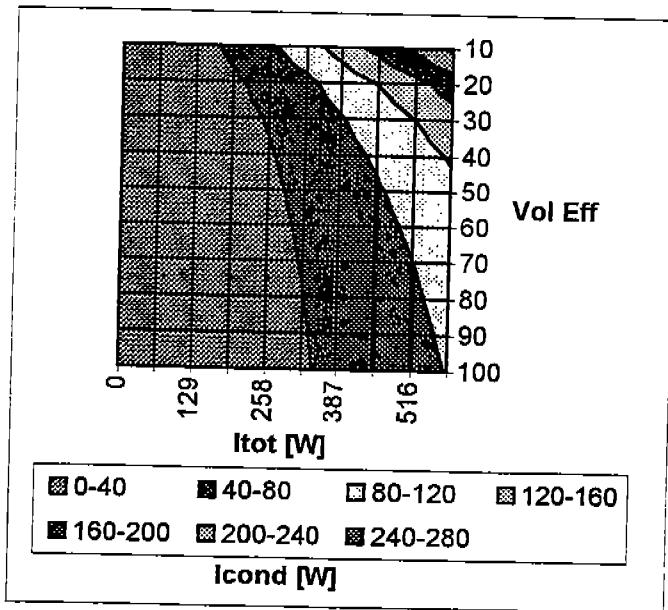


Figure 2: Internal Exergy Destruction ( $I_{tot}$ ) Vs. External Exergy Destruction ( $I_{cond}$ )

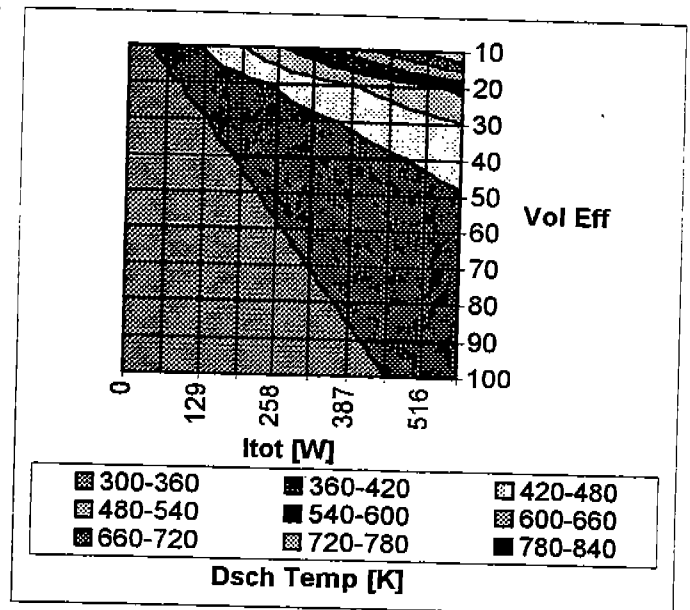


Figure 3: Discharge Temperature



Vol-eff [%]  
 $m_{ideal}$  [g/sec]  
 $m_{actual}$  [g/sec]  
 $m_{loss}$  [g/sec]  
 Is-eff [%]  
 Rat-eff [%]  
 $W$  [W]  
 $m \cdot Db$  (rat eff.) [W]  
 $I_{tot}$  [W]  
 $m \cdot Dh_s$  (is. eff.) [W]  
 $I_{tot} + I_{cond}$  [W]  
 $Dh_s$  [J/kg]  
 $Db$  [J/kg]

Initial Design	
	78.95
	19
	15
	3.96
61.69	
65.87	
743	
489	
255	
458	
286	
30544	
32616	

Table 1a

Altered Design 1/CI Volume	
	83.01
	19.00
	15.77
	3.16
61.98	
66.16	
777	
514	
264	
482	
297	
30544	
32616	

Table 2a

Altered Design 2/Friction	
	79.29
	19
	15.07
	3.885
64.12	
68.02	
718	
488	
231	
460	
259	
30544	
32616	

Table 3a

Component Losses	Exergy [W]	New Method [%]	Volumetric Eff [%]
CI Vol	0.00	$Dn_{tot-cl}$ 3.01(2.34)	$Loss_{cl}$ 7.81
Suc Thr	$I_{thr-suc}$ 76.37	$Dn_{tot-sthr}$ 11.0(8.57)	$Loss_{thr-suc}$ 7.52
Piston	$I_{thr-leak}$ 16.40	$Dn_{tot-leak}$ 2.98(2.32)	$Loss_{leak}$ 3.20
HTSuc Side	$I_{HT-suc}$ 14.02	$Dn_{tot-HTsuc}$ 2.41(1.87)	$Loss_{HT}$ 2.39
HT Mtl	$I_{HT-mtl}$ 0.29	$Dn_{tot-HTmtl}$ .03(.02)	0.00
Dsch Vlv	$I_{thr-dsch}$ 95.35	$Dn_{tot-dthr}$ 10.10(7.86)	$Loss_{thr-dsch}$ -0.06
HT Air	$I_{HT-air}$ 2.91	$Dn_{tot-HTair}$ 0.31(0.24)	0.00
HT Dsch	$I_{HT-dsch}$ 2.73	$Dn_{tot-HTdsch}$ 0.29(0.23)	0.00
Frict	$I_{fr}$ 46.07	$Dn_{tot-fr}$ 4.89(3.81)	0.00
Outside	$I_{cond}$ 31.09	$Dn_{tot-cond}$ 3.30(2.57)	0.00
Total	285.23	38.31(29.83)	20.86

Table 1b

Exergy [W]	New Method [%]	Volumetric Eff [%]
0.00	$Dn_{tot-cl}$ 1.47	$Loss_{cl}$ 3.84
$I_{thr-suc}$ 77.11	$Dn_{tot-sthr}$ 10.98	$Loss_{thr-suc}$ 7.18
$I_{thr-leak}$ 16.03	$Dn_{tot-leak}$ 2.94	$Loss_{leak}$ 3.21
$I_{HT-suc}$ 14.09	$Dn_{tot-HTsuc}$ 2.43	$Loss_{HT}$ 2.42
$I_{HT-mtl}$ 0.29	$Dn_{tot-HTmtl}$ 0.03	0.00
$I_{thr-dsch}$ 104.06	$Dn_{tot-dthr}$ 11.12	$Loss_{thr-dsch}$ -0.01
$I_{HT-air}$ 2.83	$Dn_{tot-HTair}$ 0.30	0.00
$I_{HT-dsch}$ 2.76	$Dn_{tot-HTdsch}$ 0.30	0.00
$I_{fr}$ 46.57	$Dn_{tot-fr}$ 4.98	0.00
$I_{cond}$ 32.50	$Dn_{tot-cond}$ 3.47	0.00
296.25		16.65

Table 2b

Exergy [W]	New Method [%]	Volumetric Eff [%]
0.00	$Dn_{tot-cl}$ 2.81	$Loss_{cl}$ 7.82
$I_{thr-suc}$ 76.42	$Dn_{tot-sthr}$ 11.14	$Loss_{thr-suc}$ 7.52
$I_{thr-leak}$ 16.42	$Dn_{tot-leak}$ 2.96	$Loss_{leak}$ 3.20
$I_{HT-suc}$ 13.48	$Dn_{tot-HTsuc}$ 2.20	$Loss_{HT}$ 1.97
$I_{HT-mtl}$ 0.29	$Dn_{tot-HTmtl}$ 0.03	0.00
$I_{thr-dsch}$ 95.87	$Dn_{tot-dthr}$ 10.55	$Loss_{thr-dsch}$ -0.06
$I_{HT-air}$ 2.11	$Dn_{tot-HTair}$ 0.23	0.00
$I_{HT-dsch}$ 2.67	$Dn_{tot-HTdsch}$ 0.29	0.00
$I_{fr}$ 23.33	$Dn_{tot-fr}$ 2.57	0.00
$I_{cond}$ 28.02	$Dn_{tot-cond}$ 3.09	0.00
		20.45

Table 3b

Component Losses	Exergy [%]	New Method [W]	Flow [g/sec]
CI Vol	0.00	22.35	1.485
Suc Thr	10.28	81.73	1.429
Piston	2.21	22.10	.609
HTSuc Side	1.89	17.88	.454
HT Mtl	0.04	0.23	.000
Dsch Vlv	12.84	75.00	-.012
HT Air	0.39	2.29	.000
HT Dsch	0.37	2.15	.000
Frict	6.20	36.33	.000
Outside	4.19	24.51	.000

Table 1c

Exergy [%]	New Method [W]	Flow [g/sec]
0.00	11.42	.730
9.92	85.37	1.365
2.06	22.86	.611
1.81	18.88	.459
0.04	0.24	.000
13.39	86.40	-.001
0.36	2.35	.000
0.36	2.30	.000
5.99	38.67	.000
4.18	26.99	.000

Table 2c

Exergy [%]	New Method [W]	Flow [g/sec]
0.00	20.20	1.485
10.65	79.94	1.430
2.29	21.26	.607
1.88	15.77	.375
0.04	0.23	.000
13.36	75.72	-.012
0.29	1.67	.000
0.37	2.11	.000
3.25	18.47	.000
3.90	22.18	.000

Table 3c